Displacement Ventilation

theory and design

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DISPLACEMENT VENTILATION

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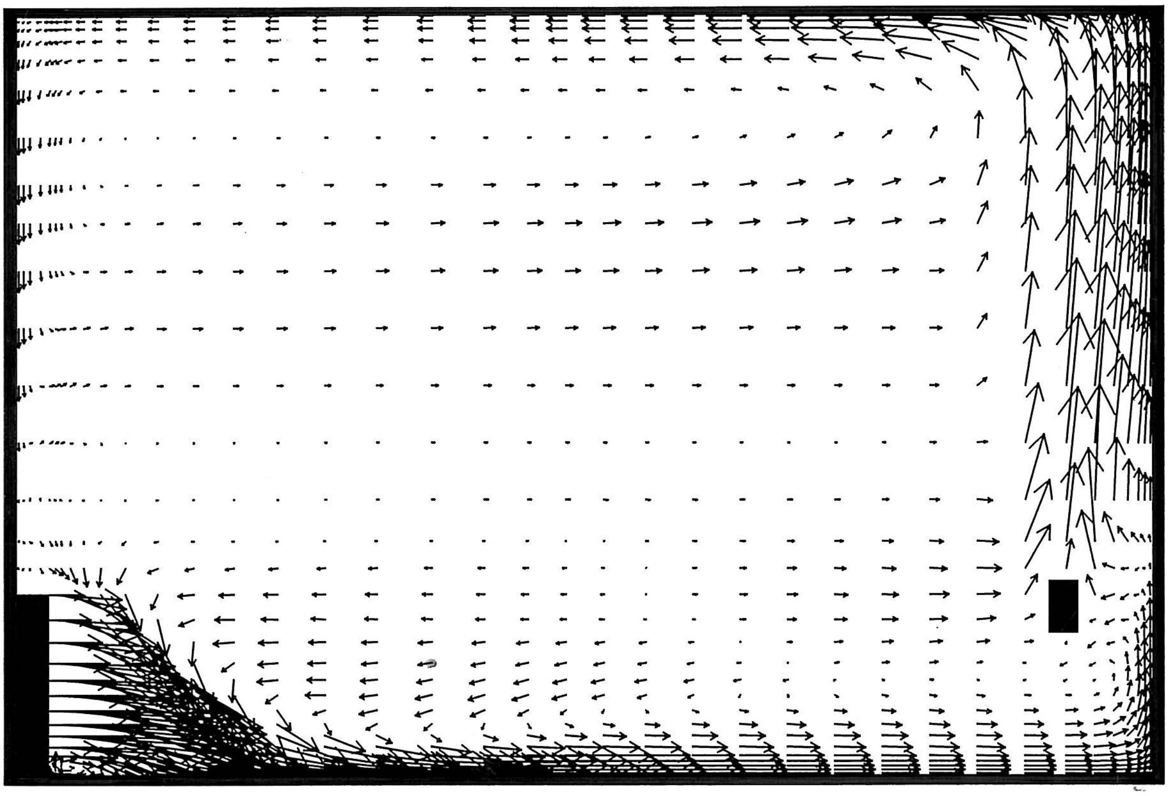

1. INTRODUCTION

For many years buoyancy driven displacement ventilation has been used in industrial areas with high thermal load as for example in hot process buildings in the steel industry. The displacement ventilation system has also grown popular during the last ten years as comfort ventilation in rooms with low thermal loads as e.g. in offices. Some main features of displacement ventilation are the possibility of creating both high temperature effectiveness and high ventilation effectiveness. The supply openings are located at a low level in case of displacement ventilation and the air flows direct into the occupied zone. Free convection from heat sources creates a vertical air movement in the room and the heated air is removed by return openings located in the ceiling or just below the ceiling. It is thus the free convection or the buoyancy which controls the flow in the room while the momentum flux from the diffusers is low and without any practical importance for the general flow in the room.

A vertical temperature gradient will arise in a room with displacement ventilation due to the vertical flow of heat to the ceiling region. It is therefore possible to remove exhaust air from the room with temperatures which are several degrees above the temperature in the occupied zone. This allows an efficient use of energy because the supply temperature can be higher than the supply temperature in case of mixing ventilation. It is, on the other hand, necessary to use a higher air flow rate in displacement ventilation to obtain a large volume in the occupied zone with fresh air and to avoid too low temperatures in the occupied zone. This will lead to a duct system which is slightly larger than the duct system used in connection with mixing ventilation.

Displacement ventilation is only used in case of cooling. It can be used as a fresh air supply system combined with a radiator heating system in cases with heating demand.

A displacement ventilation system can have high ventilation effectiveness if the sources are both heat and contaminant sources. The vertical temperature gradient implies that fresh and contaminant air are separated and the most contaminated air can be found above the occupied zone.

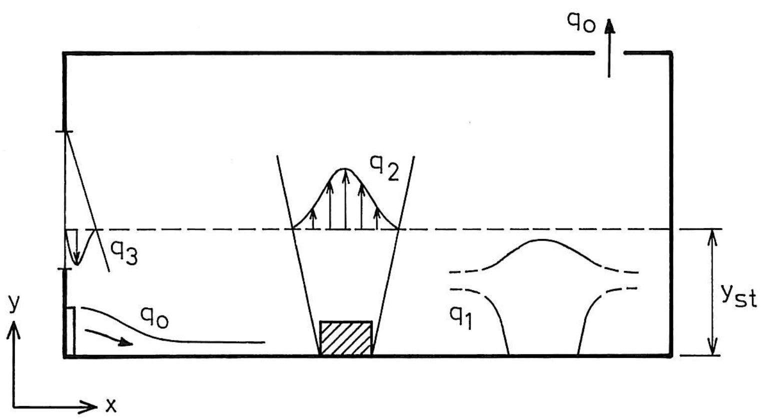

Figure 1. Room with displacement flow and natural convection.

Figure 1 shows the main principle of displacement ventilation. The airflow q_o is supplied direct into the occupied zone at low velocity from a wall-mounted diffuser. The plumes from hot surfaces, from equipment and from persons entrain air from the surroundings in an upward movement, and cold downdraft may transport air down into the occupied zone. A stratification will take place in a height where the flow q_2 - q_3 is equal to q_o in the situation shown in figure 1.

The design of a displacement ventilation system involves the calculation of the vertical flow in plumes from different heat sources. The stratification height is determined from the supply flow rate and the generated height of fresh air should be comparable with the height of the occupied zone. It is possible to estimate the vertical temperature gradient in the room. This gradient is important for thermal comfort due to the temperature difference between height of the legs and height of the head and due to radiation from the ceiling.

The diffuser can either be a wall-mounted low velocity diffuser or it can be floor-mounted diffusers. The flow from a wall-mounted diffuser moves downward to the floor due to gravity and it flows through the room in a thin layer which is typical of stratified flow, see figure 1. The flow along the floor is the only air movement which influences the comfort of the occupants and it is therefore important to have a design procedure which can support the selection of diffusers. Floor-mounted diffusers generate a high entrainment close to the openings. They should therefore mainly be distributed in secondary areas of the occupied zone.

The following chapters are arranged in the order which is used in a practical design procedure:

2. Free convection flow

3. Stratification height and concentration distribution

4. Temperature distribution

5. Velocity distribution in the occupied zone

Thermal flow from heat sources and cold surfaces can be treated independent of other flows in the room. The flow is for example independent of the stratified flow along the floor and it is often supposed to be independent of the vertical temperature gradient, although some influence is present as shown in the next chapter. A practical design procedure for displacement ventilation is often based on this simplified theory. A detailed description of the air distribution can only be achieved by full-scale experiments /1/ or by numerical prediction of the air movement as shown in references /2 and 3/.

The primary flow in a room with displacement ventilation expresses the similarity which is typical of fully turbulent flow. Vertical temperature gradients, velocity level in stratified flow at the floor, stratification level and ventilation effectiveness can all be described by an Archimedes' number independent of the velocity level in the room /4/. The Archimedes number is defined as

$$Ar = \frac{\beta g H \Delta T_o}{u_o^2} \tag{1}$$

where β, g and ΔT_o are volume expansion coefficient, gravitational acceleration and temperature difference between return and supply flows, respectively. u_o is the supply velocity and H is a characteristic height which e.g. can be the room height or the height of the wall-mounted air terminal device.

2. FREE CONVECTION FLOW

The driving force in free convection is the buoyancy effect on heated air in the room. Figure 1 shows as examples a plume from a concentrated heat source and a plume from a solar heated floor area as well as cold downdraft from a cold window surface.

2.1 Thermal Plumes from Heat Sources

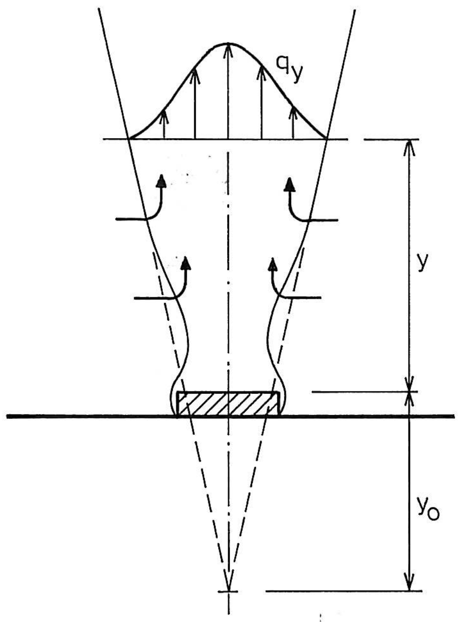

Figure 2. Free convection flow from a heat source.

Figure 2 shows the vertical flow above a heat source. The buoyancy generates a jetlike flow with a maximum velocity just above the source where the flow may expect to have a reduced diameter. Air is entrained into the plume and the width and the volume flow increase with the height.

A concentrated heat source will create a circular flow which can be expressed by

$$q_y = 0.005\ \phi_K^{1/3}\ (y + y_o)^{5/3} \tag{2}$$

where $y + y_o$ is height above a virtual origin of the flow. q_y and ϕ_K are volume flow in the height y and the convective heat emission from the source, respectively, /5/.

Equation (2) is strictly valid for $y >> d$ where d is the hydraulic diameter of the source but practice shows that the equation can be used down to a height of $y \sim 2d - 3d$, see reference /6/, which is important because the size of a heat source often can be comparable to the height of the occupied zone.

The convective heat emission ϕ_K can be estimated from the energy consumption of the heat source ϕ

$$\phi_K = k\phi \tag{3}$$

The level of the coefficient k is 0.7 - 0.9 for pipes and channels, 0.4 - 0.6 for smaller components and 0.3 - 0.5 for larger machines and components.

It can in practice be difficult to judge the location of the virtual origin but it is often assumed that $y_o \sim 2d$ for a concentrated source. The surface temperature distribution close to lighting or heating equipment may indicate rather large sources due to the radiation from the components.

A line source will create a two-dimensional flow which can be expressed by

$$q_y = 0.014\left(\frac{\phi_K}{l}\right)^{1/3} (y + y_o)l \tag{4}$$

where l is the length of the source /7/. y_o corresponds to 1 - 2 times the width of the heat source.

A heat source in the occupied zone may also be the contaminant source in the room. Typical examples are printers in offices, welding plumes in industrial areas and people in general. The volume flow q_y is therefore the contaminated part of the flow in the room and the concentration distribution across the flow is distributed in a similar manner as in the velocity profiles indicated in figure 1.

Equations (2) and (4) are only strictly valid for homogeneous surroundings. The flow in surroundings with a vertical temperature gradient will show a decreasing velocity level compared to homogeneous surroundings because the temperature gradient will diminish the buoyancy force.

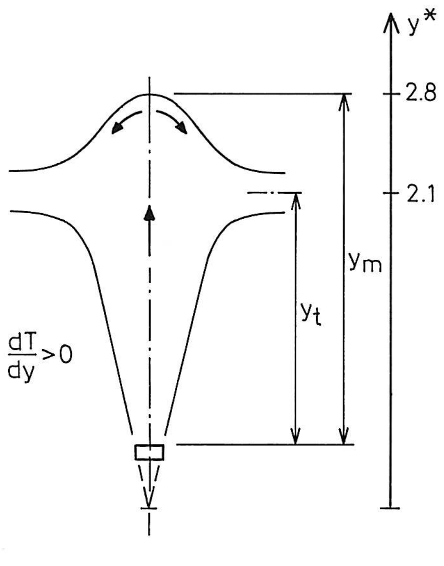

Figure 3. Vertical plume in a room with temperature gradients and stratification.

Figure 3 shows the flow from a heat source in a room with a strong temperature gradient. The flow will be buoyancy neutral in the height y_t but it will continue to the height y_m due to the momentum in the plume. The flow will finally terminate in a stratified layer at y_t.

The convective flow can be calculated from the following model given by Morton et al. /8/ and addressed by Mundt in /9/.

A dimensionless height y^* is calculated for different heights y above the source

$$y^* = 2.86(y + y_o)\frac{dT}{dy}^{3/8} \phi_K^{-1/4} \tag{5}$$

where dT/dy is the vertical temperature gradient. Figure 3 shows that only y^* values less than 2.1 are relevant to further calculations. A dimensionless parameter m is given by

$$m = 0.004 + 0.039y^* + 0.380y^{*2} - 0.062y^{*3} \tag{6}$$

and the volume flow rate in the height y^* is given by

$$q_y = 2.38 \cdot 10^{-3} \cdot m \cdot \phi_K^{3/4} \cdot \left(\frac{dT}{dy}\right)^{-5/8} \tag{7}$$

The maximum height y_m is given by equation (5) for $y^* = 2.8$

$$y_m = 0.98\,\phi_K^{1/4}\left(\frac{dT}{dy}\right)^{-3/8} - y_o \tag{8}$$

and the height for neutral buoyancy and stratification of the plume is given by

$$y_t = 0.74\ \phi_K^{1/4}\left(\frac{dT}{dy}\right)^{-3/8} - y_o \tag{9}$$

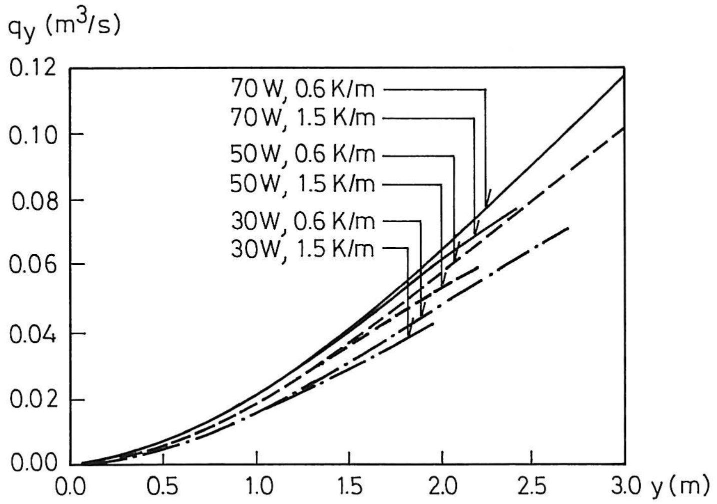

Figure 4. Free convection flow above a heat source of 30, 50 and 70 W in a room with different gradients. Reference /9/.

Figure 4 shows the flow in plumes from different sources. It is obvious that the flow rate is only moderately influenced by the temperature gradient while the maximum height y_m is strongly influenced by the temperature gradient. This is an important aspect in e.g. a workshop with welding because the welding fume will be concentrated in height y_t in case of a temperature gradient. It can therefore be concluded that the temperature gradient dT/dy should be small but, on the other hand, sufficient to create the stratification which is the background for displacement ventilation and for a clean occupied zone.

2.2 Flow above People and Components

Many practical heat sources are large compared to room dimensions and it is difficult to estimate the type of plume (circular or plane). Furthermore, it is difficult to judge the convective heat emission ϕ_K and the distance to the virtual origin y_o and it may be difficult to use the equations (2), (4) and (7) close to the equipment. A logical conclusion is to measure the volume flow above a number of typical sources in rooms as e.g. a person, office equipment, lighting equipment and catering equipment as described in the following.

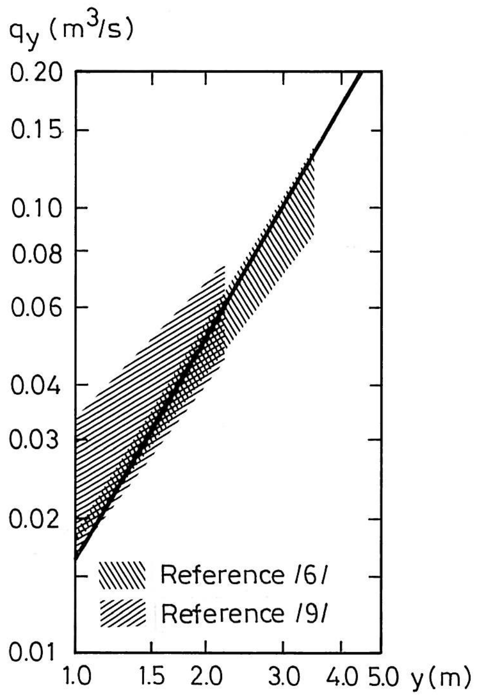

Figure 5. Free convection flow above a sedentary person. ϕ = 100 W. The height y is measured from floor level.

Figure 5 shows the flow above a source simulating a sedentary person. Measurements by Kofoed and Nielsen /6/ show influence from the vertical temperature gradient. Measurements in the upper part of the shaded area correspond to 0.09 K/m while the lower part of the area corresponds to 0.3 K/m. The measurements discussed by Mundt /9/ are a joint project between a number of manufacturers in the Scandinavian countries. The measurements are made for temperature gradients between 0.6 K/m and 1.5 K/m which are typical levels for displacement ventilation. The full-drawn line on figure 5 shows the volume flow according to equation (2) for $k = 1/3$, $y_o = 0$ and y measured from floor level. Measurements by Fitzner /10/ show similar results as well as a strong decrease in entrainment at increasing temperature gradient. This effect is especially pronounced in heights close to neutral buoyancy.

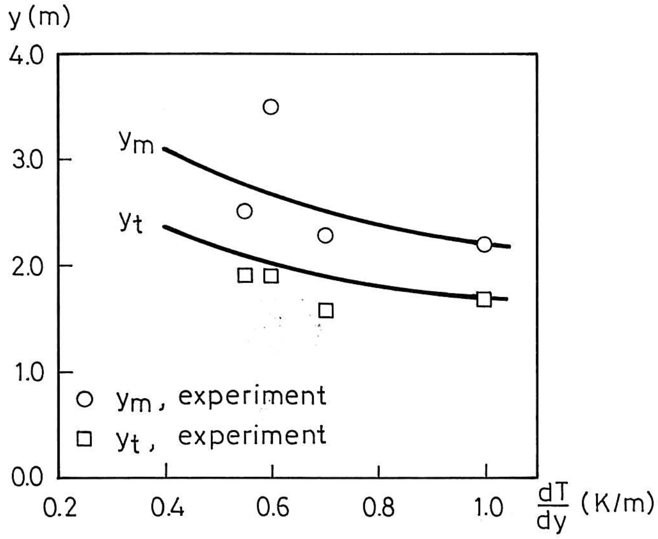

Figure 6. Maximum height and height to neutral buoyancy in a plume above a sedentary person. $\phi = 75$ W. The height y is measured from floor level $y_o = 0$ and $k = 1/3$.

Figure 6 shows measurements of maximum height and height to neutral buoyancy in a plume above a source simulating a sedentary person. y_m and y_t are also given according to equations (8) and (9).

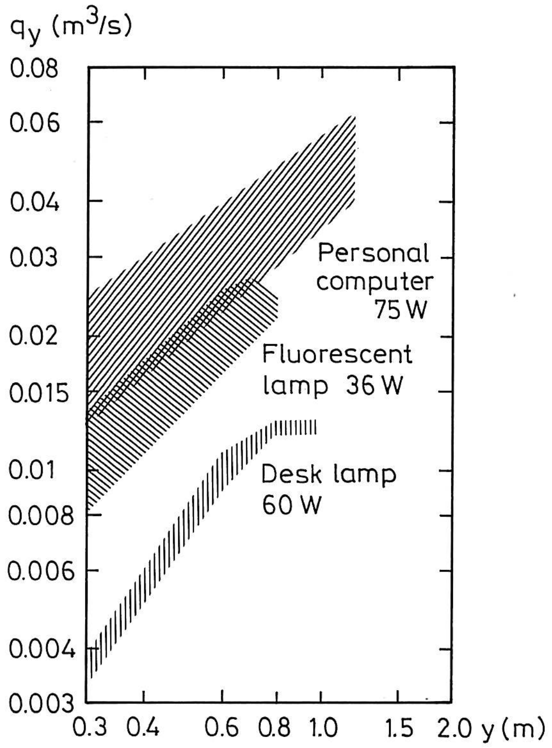

Figure 7. Free convection flow above a PC and different lamps.

Figure 7 shows the measurements given by Mundt /9/. It should be noticed that there is a large difference between the volume flow from the computer and the flow from a desk lamp, although the energy consumption ϕ is at the same level. The shaded areas show the distribution of the measurements for different temperature gradients. The height to neutral buoyancy must be found from equation (9).

2.3 Interaction between Thermal Plumes and Influence from Walls

The plume from a heat source located close to a wall may be attached to the wall due to the Coanda-effect, see figure 8A. The entrainment will be reduced compared to the entrainment in a free plume and this will influence the location of the stratification height in the room.

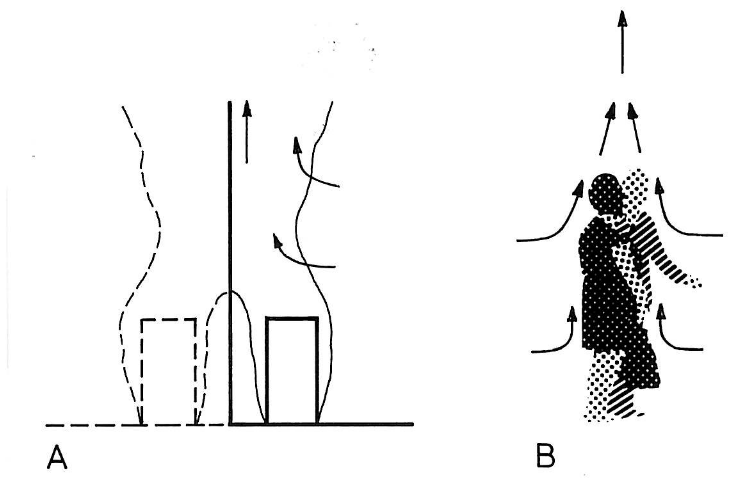

Figure 8. A) Thermal plume attached to a wall. B) Interaction between two thermal plumes.

The flow in a plume close to a wall is similar to one half of the flow in a free plume with a convective heat emission of 2 ϕ_K as indicated on the figure. The flow in the boundary layer close to the wall has a minor influence on the general air movement, and the volume flow q_{yw} in the actual plume is given from equation (2) for the heat emission of 2 ϕ_K and an entrainment of 2 q_{yw}.

$$q_{yw} = 0.0032\,\phi_K^{1/3}(y + y_o)^{5/3} \tag{10}$$

The equation shows that the flow rate is 63% of the flow in a free plume.

It can be shown that the flow is reduced to 40% of the flow in a free plume if the heat source is located close to a corner in the room. The flow rate q_{yc} in the plume is given from equation (2) for the heat emission of $4\phi_K$ and an entrainment of $4q_{yc}$

$$q_{yc} = 0.002\,\phi_K^{1/3}(y + y_o)^{5/3} \tag{11}$$

The thermal flows above a number of sources may be absorbed into one plume due to the Coanda-effect if the sources are located close to each other. The flow q_{yN} from N identical sources is given by

$$q_{yN} = N^{1/3}\, q_y \tag{12}$$

where q_y is the flow in a plume from one of the sources. The flow above the two persons on figure 8 B will therefore be 1.26 times the flow above a single person at equivalent heights. It is obvious from the equation that the flow in a merged plume from a high number of heat sources is small compared to the flow in individual plumes with sufficient space for entrainment.

Measurements in plumes close to a wall or a corner are reported by Kofoed and Nielsen /11/ and they support the assumption behind equation (10) and (11).

2.4 Thermal Flow and Cold Downdraft from Vertical Surfaces

A thermal flow will be generated close to a surface if the surface temperature is different from the air temperature.

A cold downdraft will take place at a cold surface. The air close to the surface will be cooled and it will obtain an increased density which will move the air downward due to the gravity force. Any heat loss through a vertical surface in areas with low velocities is connected with cold downdraft.

The volume flow in a cold downdraft or in a rising thermal flow close to a warm surface is given by the following equation, reference /12/

$$q_y = 2.8 \cdot 10^{-3}\, |\Delta T|^{2/5}\, y^{6/5}\, l \tag{13}$$

where q_y is the flow in the turbulent free convection boundary layer of the height y measured from the upper edge or lower edge of the surface. ΔT is the temperature difference between room temperature and surface temperature and l is the horizontal width of the surface.

Figure 1 shows a cold downdraft at a window. It appears that the volume flow q_3 will help to increase the stratification height. The flow in cold downdraft will also transport contaminant from upper zone down into the lower zone which is a problem in cases where the displacement ventilation is used for contaminant control.

The flow from cold downdraft may continue through the occupied zone causing discomfort due to the velocity of the flow. This effect will be discussed in the chapter on velocity distribution in the occupied zone.

Displacement ventilation is often connected to a large vertical temperature gradient. Heat losses may therefore take place in the upper part of the room and heat gains make likewise take place in the lower part of the room. A downward boundary layer flow will be generated in the upper part of the room and an upward flow will be generated in the lower part of the room, all with velocity levels dependent on the actual level of the heat fluxes.

3. STRATIFICATION HEIGHT AND CONCENTRATION DISTRIBUTION

It is necessary to consider both air quality and thermal comfort in a room with displacement ventilation. Air quality is an important aspect in case of contaminant emission in the room and thermal comfort is a general aspect in connection with design of a system. The air quality aspect will be considered in this chapter in connection with determination of stratification height, concentration distribution and ventilation effectiveness, and thermal comfort aspects will be considered in the last two chapters in connection with the determination of temperature distribution in the room and velocity distribution in the occupied zone.

3.1 Stratification Height

The volume flow to a room is calculated from the stratification height as explained on figure 1.

The heat source in the room in figure 1 will generate a plume. This plume will entrain a flow of q_2 up to the stratification height y_{st} which corresponds to the supply flow q_o and the cold downdraft q_3. The plume will continue above the stratification height and the entrainment in the upper zone will mix the air in this area.

The heat source in the room can also be a contaminant source. The contamination will move upward in the plume and it will mix in the upper zone. The stratification height is therefore the height of a lower zone which mainly contain fresh air from the supply system.

The supply flow rate q_o is connected to q_2 and q_3 via

$$q_o = q_2 - q_3 \tag{14}$$

where the flows q_2 and q_3 are obtained as described in the last chapter for a height which corresponds to the design value of y_{st}.

The thermal flow q_1 on figure 1 will not have any influence on the stratification height because the height to neutral buoyancy is smaller than y_{st}.

The statification height is influenced by the flow rate q_o. An increase in q_o will increase the height y_{st} because there will be more primary air in the occupied zone for entrainment into the plume. An increase of the supply air q_o will also decrease the temperature difference $T_R - T_o$ between the return and supply temperature and it is important that this difference is sufficient for the maintenance of stratification in the room.

The cold downdraft shown in figure 1 will increase the stratification height and it will also make a transport of contaminant from the upper zone into the room down into the lower zone.

Displacement ventilation requires a height volume flow rate to the room. Figure 5 indicates that the basic ventilation requirement of 0.01 m^3/s per person will maintain a stratification level of about 0.75 m and this height is hardly sufficient if the effect of a lower zone with fresh air is the main feature in the design of the system.

A stratification height of minimum 1.1 m can be recommended in rooms where the occupants mainly have a sedentary working position. Figure 9 shows that the free convection around a person will generate a flow of clean air in the breathing zone although the stratification height is smaller than the height to the breathing zone in undisturbed surroundings. The measurements shown in figure 9 are made by Stymne et al. /13/ and they indicate that the lower zone can be locally displaced approximately 0.2 m upward around heated bodies.

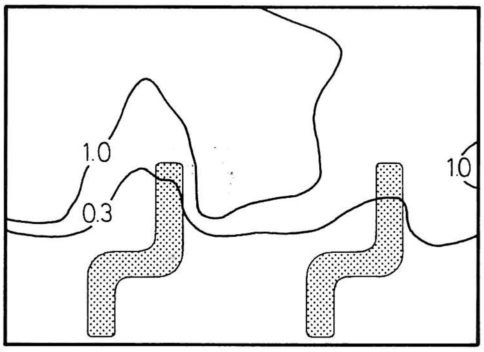

Figure 9. Normalized concentration distribution c/c_R around two persons. The ventilation rate is 0.011 (m^3/s, person) and the source of tracer gas is located in the upper zone close to the ceiling.

The air movement in the upper zone is often rather well mixed which means that the normalized concentration c/c_R, where c_R is the concentration in the return opening, is close to 1.0. A normalized concentration distribution of 1.0 is the concentration distribution which can be obtained by a mixing air distribution system and it can therefore be concluded that even cases with 1.0 concentration in the breathing zone should be acceptable in certain situations.

The stratification height in an industrial environment must be higher than the height of the occupied zone if the sources are combined heat and contaminant sources, and if the contaminant constitutes any health risk. Only measurements in the workers' breathing zone can demonstrate if the system works properly with respect to air quality.

3.2 Concentration Distribution

Figure 10 shows the vertical concentration distribution in a room where the heat source also works as a contaminant source. The location of the stratification height is clearly indicated by the measurements and both the lower and the upper zone have a well mixed concentation distribution. The tracer gas concentration in the upper zone is close to the concentration in the return opening, c_R, and the concentration in the lower zone is 0.1 to 0.3 times c_R which is typical of many situtations in highly loaded rooms.

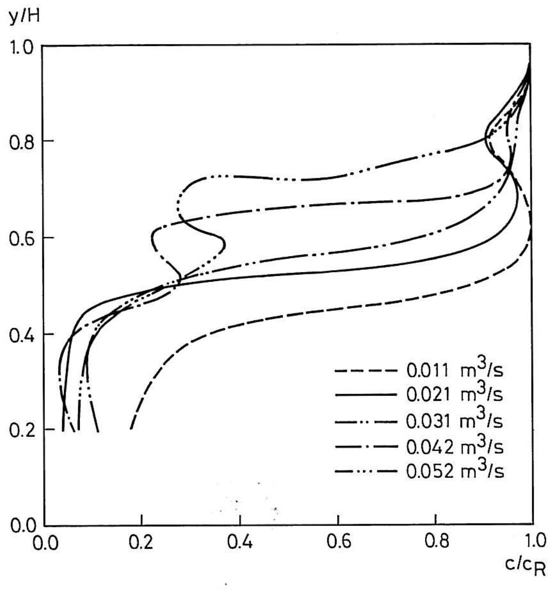

Figure 10. Vertical concentration distribution in a room with a combined heat and contaminant source versus volume flow rate to the room. Reference /14/.

The figure shows that the stratification height y_{st} is a function of the volume flow rate q_o to the room. This is also obvious from equation (2) which indicated that y_{st} is proportional to $q_o^{3/5}$ at constant heat load. It is further possible to show that the stratification height is a unique function of the Archimedes number and proportional to $Ar^{-1/5}$ in a geometry with concentrated heat sources.

Two of the concentration profiles on figure 10 show a small concentration peak just below the main stratification height. This increase is generated by cold downdraft at the wall which has a neutral buoyancy in the height $y/H \sim 0.5$.

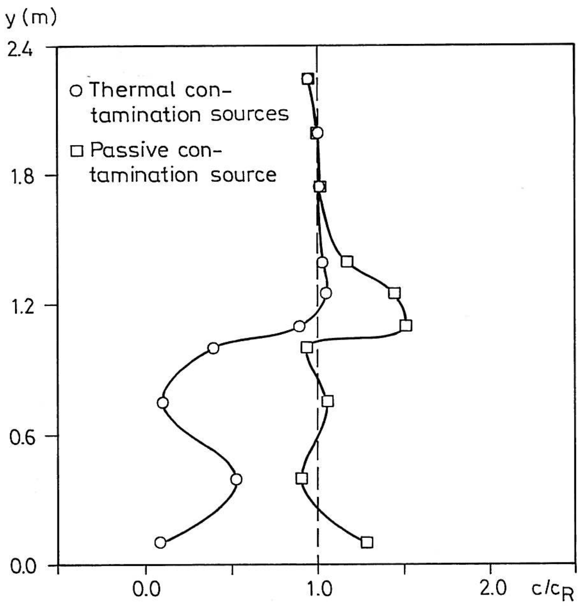

Figure 11. Vertical contaminant distribution in a room with displacement ventilation and two locations of the contaminant source, reference /1/.

The relative concentration distribution given on figure 11 shows in one example a stratification with a high concentration in the upper area of the room and a low concentration in the lower part of the room. The heat load consists of four thermal manikins and the tracer gas is released in the plumes from the manikins. The average value of the concentration in the lower part of the room is 0.25 c_R. The situation is different when the contaminant source is passive and located outside the plumes below the stratification height. The figure shows a high level of concentration in the room height corresponding to the height of the source. Very high concentrations may be obtained in this situation because the air movement is small in the area around the source and the contaminant is stabilized in a horizontal layer due to the temperature gradient.

The low velocity level and the vertical temperature gradient in a room with displacement ventilation will relaminarize the air movement. The consequence for a passive contaminant source is a high concentration level as indicated on figure 11. The free convection flow around a person may protect the breathing zone from surrounding contamination at the head level as shown on figure 9, but it may also attract contaminant from a passive source with a vertical location in the room below the breathing zone of the person. This process has been measured by Holmberg et al. /15/ and it is indicated on figure 12.

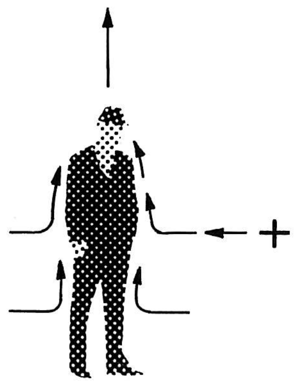

Figure 12. Contaminant transport from a passive source to the breathing zone of a person.

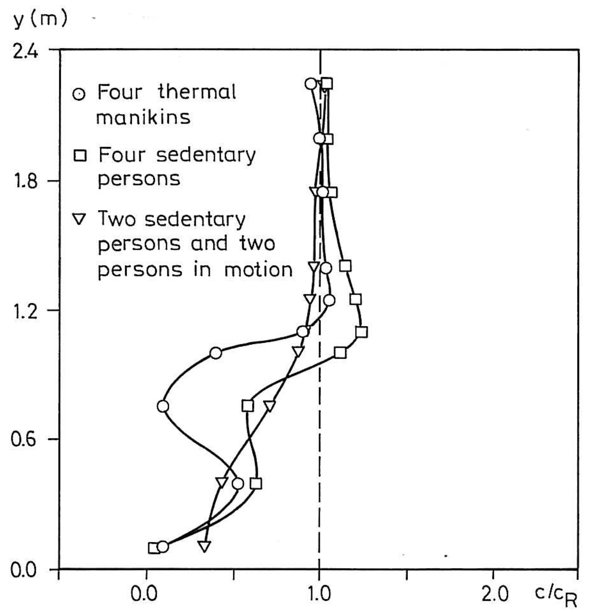

Figure 13. Concentration distribution in a room with thermal manikins, sedentary persons and persons in motion, reference /1/.

It is important to preserve the stratification in the room when persons are present and in motion. Figure 13 shows the vertical concentration distribution with four thermal manikins. A CO_2 tracer gas is released in the plumes above the manikins. The relative concentration distribution for four persons is also shown and the CO_2 concentration is, in this case, the values obtained from the presence of persons in the room. It is shown that two persons in motion are able to smooth the vertical gradient slightly, but it is in all situations possible to observe a stratification of CO_2.

3.3 Ventilation Effectiveness

The air quality and the efficient use of air are as important as thermal comfort. Different definitions of effectiveness for the evaluation of an air distribution system have therefore been commonly used during recent years.

The ventilation effectiveness shows how fast contaminant is removed from a room. It is defined as the ratio of concentration of the contaminant in the return opening to the concentration of contaminant in areas of the ventilated room.

The ventilation effectiveness ε_{oc} of the occupied zone is given by

$$\varepsilon_{oc} = \frac{c_R}{c_{oc}} \tag{15}$$

where c_R and c_{oc} are concentration in return opening and mean concentration in the occupied zone, respectively.

A local ventilation index ε_P is defined as

$$\varepsilon_P = \frac{c_R}{c_P} \tag{16}$$

where c_P is the concentration in a point of the room, e.g. the breathing zone of a person. ε_P is also the reciprocal of the normalized concentration in the point P.

A mean ventilation effectiveness $\bar{\varepsilon}$ is given by

$$\bar{\varepsilon} = \frac{c_R}{\bar{c}} \tag{17}$$

where $\bar{c}$ is the mean concentration in the whole room including areas outside the occupied zone.

Equations (15) to (17) assume that the supply flow is uncontaminated and it is also assumed that both the contaminant process and the flow are steady.

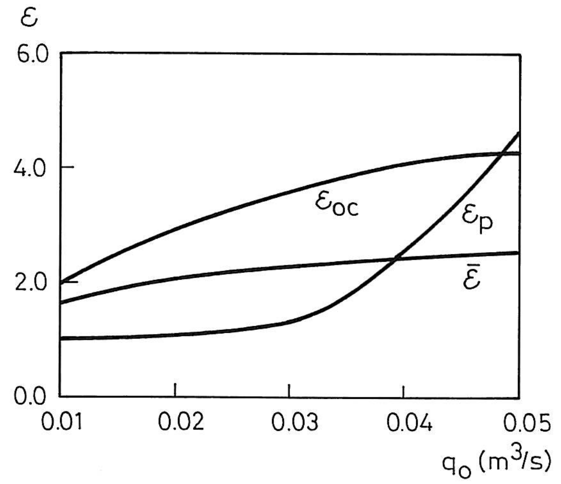

Figure 14. The ventilation effectiveness $\bar{\varepsilon}$ and ε_{oc} as well as the ventilation index ε_P for the measurements shown on figure 10.

Figure 14 shows the level of information which can be obtained from measurements of different versions of the ventilation effectiveness. The mean ventilation effectiveness $\bar{\varepsilon}$ indicates a low level at all air flow rates and the level is strongly influenced by the contaminant in the upper zone of the room. The ventilation effectiveness of the occupied zone ε_{oc} has a higher level because c_{oc} is calculated up to the height y/H = 0.72 which excludes the influence of a large part of the upper zone. The local ventilation index is measured in the height y/H = 0.6 in the case shown in figure 14. This index is very sensitive to the flow rate to the room because the height of the stratification crosses the measuring area at the actual flow rate.

It is typical that the ventilation effectiveness is larger than 1.0 in many practical situations with displacement ventilation and this is different from the situation in a room with conventional mixing ventilation.

It is easy to measure the mean ventilation effectiveness $\bar{\varepsilon}$ in small rooms because, in principle, it can be made by two concentration measurements. The tracer gas concentration in the return opening c_R is measured during the steady state of an experiment. The mean concentration $\bar{c}$ of the room is measured by mixing the room air by a fan immediately after shut down of ventilation system and tracer gas emission. The measurements of the ventilation effectiveness ε_{op} are more demanding because it is necessary to have a number of measuring points in the occupied zone to obtain the mean concentration in that area. It is of course easy to measure the ventilation index ε_P because it only involves two measurements in a steady state condition in the room.

The ventilation effectiveness can be defined from the height of the stratification y_{st} due to the presence of stratified flow.

$$\bar{\varepsilon} = \frac{H}{H + y_{st}\,(c_1/c_R - 1)} \tag{18}$$

It is assumed that the air in the lower and the upper zone is well mixed separately with a concentration of c_1 in the lower zone and a concentration in the upper zone which is equivalent to c_R

The ventilation effectiveness ε_{oc} of the occupied zone is given by

$$\varepsilon_{oc} = \frac{c_R}{c_1} \quad \text{for} \quad y_{oc} \leq y_{st} \tag{19}$$

and

$$\varepsilon_{oc} = \frac{y_{oc}}{y_{oc} + y_{st}\,(c_1/c_R - 1)} \quad \text{for} \quad y_{oc} > y_{st} \tag{20}$$

where y_{oc} is the height of the occupied zone.

4. TEMPERATURE DISTRIBUTION

A displacement ventilation system has an efficient use of energy because it is possible to remove exhaust air from the room where the temperature is several degrees above the temperature in the occupied zone which will allow a higher air inlet temperature at the same load.

The temperature distribution will also influence the thermal comfort in the room. It is therefore necessary to consider the consequences of a temperature gradient in the occupied zone as well as the asymmetric radiation from the ceiling.

4.1 Vertical Temperature Gradient for Different Types of Thermal Loads

The vertical temperature gradient is very characteristic of a room with displacement ventilation. Heat from heat sources is supplied to the room as convection and radiation. Free convection will raise the ceiling temperature compared to the surroundings, and radiation from the ceiling will then increase the temperature of the floor which, on the other hand, is cooled by the cold supply flow from the diffuser. The total effect is a vertical temperature gradient which is rather similar at different locations due to the stratification in the room.

The temperature gradient can be given with advantage in a dimensionless form. Figure 15 shows that the gradient has a limited variation compared with the variations which will be found for gradients in a dimensioned form.

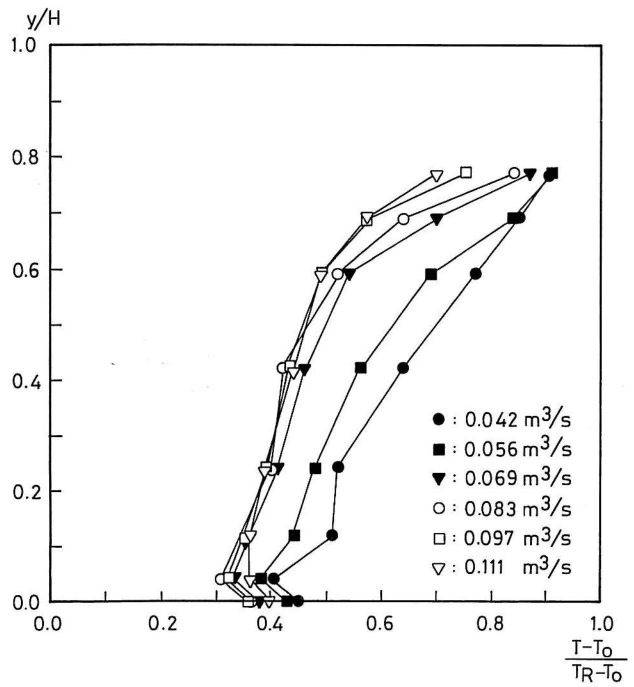

Figure 15. Vertical temperature distribution for different air flow rates, reference /16/.

The measurements shown on figure 15 are all made at constant heat load from a concentrated heat source. A closer examination of the measurements shows that an increasing Archimedes' number will decrease the normalized gradient slightly, while the dimensioned gradient will increase at increasing Archimedes' number. The dimensionless temperature gradient is a unique function of the Archimedes number when the flow is a high turbulent flow and experiment in /4/ indicates this assumption in practice although some areas of a room may have an air movement with relaminarization.

The vertical temperature gradient is independent of the location of the heat load in the room as long as the elevation of the source is constant. The gradient is, on the other hand, strongly influenced by variation in elevation and the most energy efficient layout is a design with a high location of the heat sources. The ultimate use of this principle is to combine the ceiling lighting with the exhaust openings.

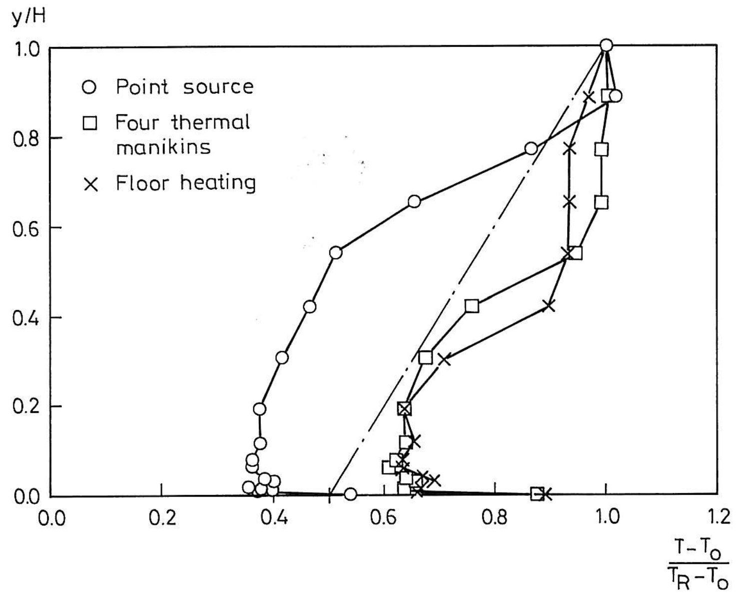

Figure 16. Vertical temperature gradients in a room with different heat sources at equal Archimedes' number /1/.

Figure 16 shows the vertical temperature gradient for different heat sources. The concentrated heat source is a small cylindrical heater with open heating elements, $0.3 \times 0.1^{\varnothing}$ m. The thermal manikin is a black painted cylinder with the dimensions $1.0 \times 0.4^{\varnothing}$ m. The floor heating consists of more electrical heating carpets covering a large part of the floor surface.

The location of the normalized temperature gradient on figure 16 depends on geometrical extension and temperature of the heat source. A heat source as the concentrated source will give a temperature distribution with high system effectiveness, while four thermal manikins will generate a temperature distribution with lower effectiveness. Floor heating shows a bad utilization of displacement flow. It is likely that the ratio between radiation and convection is an important parameter. A high level of this ratio will displace the curves to the right side on figure 16 because it will increase the amount of heat supplied to the floor. Experiments with four thermal manikins covered with aluminium foil support this theory because the vertical temperature profile in this situation is displaced to the left side on figure 16.

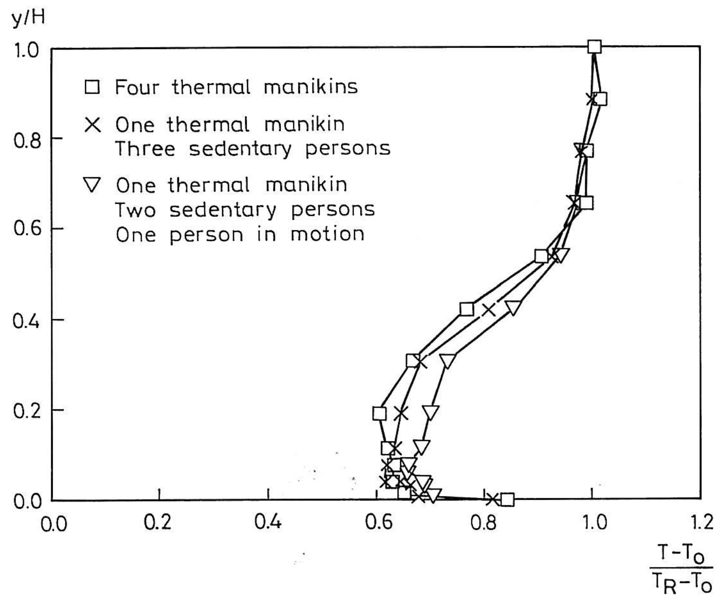

Figure 17. Vertical temperature gradients in a room with thermal manikins, sedentary persons and persons in motion /1/.

Figure 17 shows the vertical temperature distribution in a room with thermal manikins and persons. The manikins seem to give a sufficient thermal description of a person. It is especially important to see that a person in motion is unable to spoil the stratification, and the measurements show only a slight reduction in the effectiveness of the system. Other measurements with heavy activity and an open door to the test room do also confirm the large stability of the stratified flow in the room.

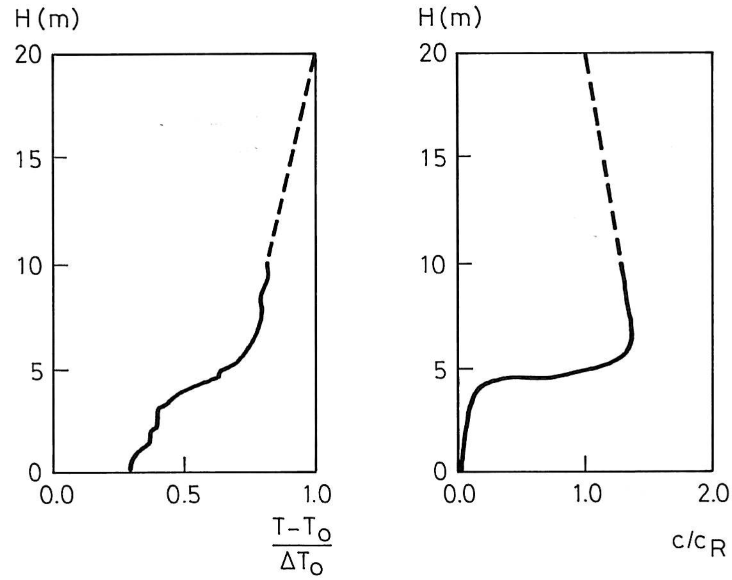

Figure 18. Temperature and concentration distribution in an industrial building.

The profiles shown on the figures 15 to 17 are made in a room of conventional size (H ~ 2.5 m). Most of the experience with large rooms comes from measurements in industrial areas and figure 18 shows the temperature distribution and the concentration distribution measured in a silicon carbide furnace room, Skistad /17/.

The figure shows a moderate change in temperature gradient and a very steep change in concentration gradient in the stratification height. Both gradients behave more or less in the same way as found in smaller rooms.

The contaminant is carbon monoxide from the furnaces. It is seen that the air with the high concentration level must be a convection flow from parts of a furnace with a lower temperature level than the surface temperature which generates the maximum temperature below the ceiling.

It has to be considered that it is difficult to obtain steady state solutions when measurements are made in large areas due to the large air volume and thermal mass. It is possible to use this effect in ventilation of areas which have a high level of the loads in short periods during the day.

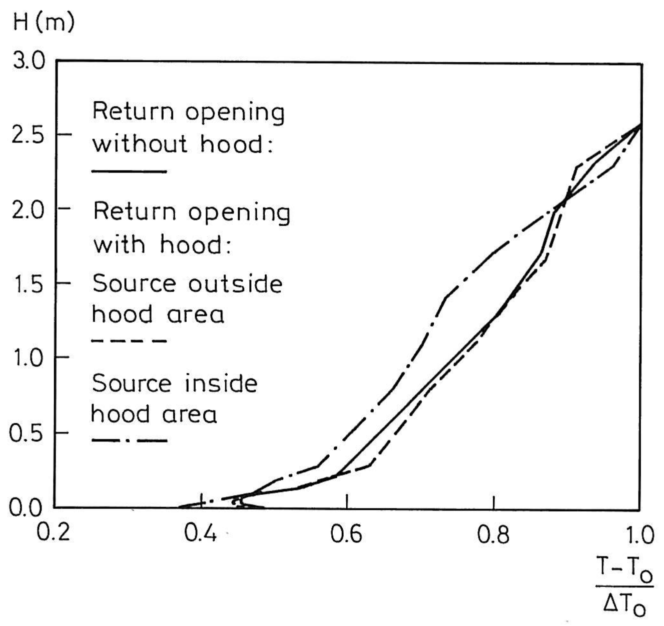

Figure 19. Experiments with return opening geometry.

It is generally assumed that the location of the return opening is of minor importance if it is located just below the ceiling or in the ceiling, because the air in the upper zone is almost fully mixed.

Experiments show that there is some influence from the details around the opening. A heat source with a plume which reaches the ceiling is ventilated in an efficient way if the opening is located above the source and if a small hood is located around the opening. Figure 19 shows some experiments with different location of heat source and return opening. A short hood or shield with a height of 0.6 m around the return opening will increase the efficiency of the opening and the measurements show that the return temperature will increase when the plume reaches the ceiling with the centre line inside the hood. High flow rates in the room may destroy the effect.

4.2 Design Temperature Gradient

A design of a displacement ventilation system involves the prediction of the vertical temperature gradient. Knowing this gradient it is possible to calculate the thermal sensation of the temperature level, the thermal sensation of the temperature gradient and the thermal sensation of the asymmetric radiation from the ceiling.

Measurements indicate that it is possible to make the simplified assumption that the temperature varies linear with the height from a minimum temperature at floor level T_f to a maximum temperature at ceiling level which is equivalent to the return temperature T_R.

$$T = \frac{y}{H}\left(T_R - T_f\right) + T_f \qquad (21)$$

Skistad /17/ assumed that $T_f - T_o$ is equivalent to 0.5 $\left(T_R - T_o\right)$ in all practical situations. Comparison with figure 15 shows that the minimum temperature T_f has a limited variation when it is given in a dimensionless form and this may support the above-mentioned assumption. The dotted line on figure 16 shows the variation of T for $T_f - T_o = 0.5\ \Delta T_o$ and it can be argued that the line represents some mean assumption for gradients from different types of heat sources.

Figure 15 shows that $\left(T_f - T_o\right)/\Delta T_o$ varies slightly with the air flow rate. Mundt addresses this effect in /18/ and gives the following figure 20 for the variation as a function of the air flow rate per m² floor area. The figure summarize a large number of measurements in different rooms.

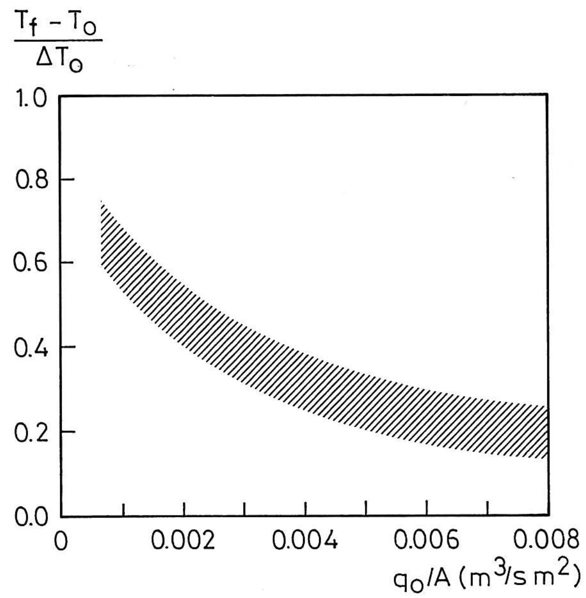

Figure 20. Minimum temperature at floor level T_f in a room with displacement ventilation versus air flow rate per square meter floor area.

4.3 Temperature Effectiveness

The efficient use of air in a ventilation system can be studied by the temperature effectiveness ε_T of the occupied zone.

$$\varepsilon_T = \frac{T_R - T_o}{T_{oc} - T_o} \tag{22}$$

where T_R, T_o and T_{oc} are temperature in return opening, temperature in supply opening and mean temperature in the occupied zone, respectively.

It is possible to express the temperature effectiveness ε_T by

$$\varepsilon_T = \Delta T_o \Bigg/ \left(\frac{y_{oc}}{2H} \left(T_R - T_f\right) + T_f - T_o \right) \tag{23}$$

if it is assumed that the vertical temperature distribution in the room is linear.

Figure 21 shows the variation of the temperature effectiveness as a function of the Archimedes number. The measurements are given on figure 15 and it is obvious that the temperature effectiveness is a unique function of the Archimedes number independent of flow rate and heat load. Two different types of diffusers are used in the experiments and the figure shows that the temperature effectiveness seems to be uninfluenced by the type of diffuser selected.

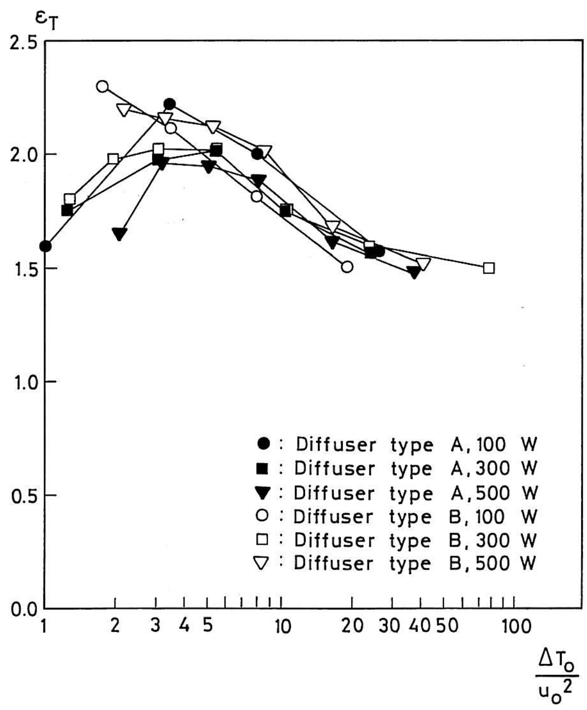

Figure 21. Temperature effectiveness versus the Archimedes number. The Archimedes number is expressed as $\Delta T_o / u_o^2$ (Ks^2/m^2) and the height of the occupied zone is measured to $y_{oc}/H = 0.81$.

The effectiveness of the air distribution system can also be expressed by a load factor. If, for example, the heat is generated below the ceiling it will contribute with a small load on the air conditioning system compared to heat generated in the occupied zone and this effect can be expressed by the following load factor

$$\mu_T = \frac{T_{oc} - T_o}{T_R - T_o} \tag{24}$$

The load factor μ_T is the reciprocal of the temperature effectiveness ε_T.

Figure 22 shows load factors for different locations of heat sources for ventilation in a room with a number of vertical jets in the floor. It is shown that a heat source at the floor, as for example solar radiation, results in a temperature in the occupied zone close to the return temperature which gives a high load factor. Load from people and equipment at table level and solar radiation on a wall will give a small load factor, while heat sources at higher level, as lighting in the ceiling, result in a reduction of the load on the air conditioning system, see Detzer and Jungbäck /19/. The lighting will heat the surfaces in the room due to radiation and this will limit the possibility of making reductions in the load factor below certain limits.

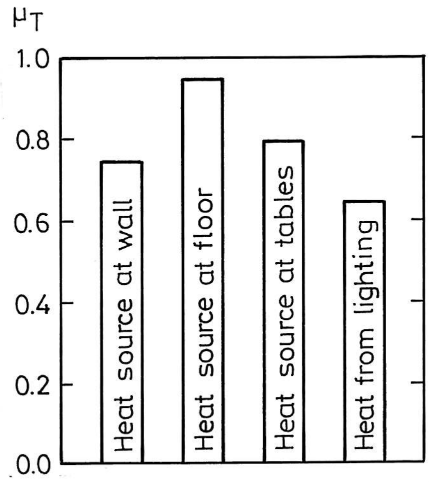

Figure 22. Load factors for different locations of heat sources in a room with floor-mounted diffusers.

5. VELOCITY DISTRIBUTION IN THE OCCUPIED ZONE

The air terminal devices for displacement ventilation are located in the lower part of the room. The airflow is supplied direct into the occupied zone and it is therefore important to have a design method which can predict the velocity distribution close to the floor. The velocity level outside the floor areas is very small due to the stratification and the sensation of draft is therefore primarily connected to the flow at floor level.

5.1 Velocity Distribution in the Flow from a Wall-Mounted Diffuser

Figure 23 shows the velocity distribution in front of a wall-mounted low velocity diffuser. The flow has a radial distribution when it leaves the diffuser and it will accelerate towards the floor due to the gravity effect on the cold air. The velocity contour for the 0.2 m/s velocity indicates an area close to the diffuser where the velocity is too high for a sedentary person.

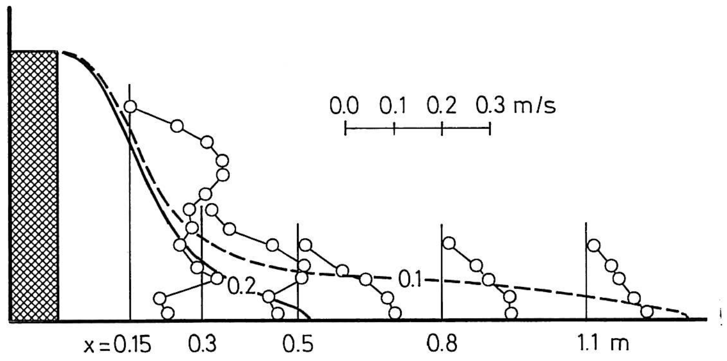

Figure 23. Velocity profiles and velocity contours in front of a wall-mounted low velocity diffuser type G. q_o = 0.029 m^3/s and T_{oc} - T_o = 6 ^oC.

The design of a displacement ventilation system with a wall-mounted diffuser involves the determination of a length l_n which is the distance from the diffuser to the 0.2 m/s velocity contour measured along the centre line of the flow. The distance l_n is a function of the flow rate from the diffuser and it is a function of the temperature difference T_{oc} - T_o. It is important that l_n is small compared to the length of the room because it corresponds to an area which may be strongly influenced by draft.

Figure 23 shows that l_n is 0.5 m at the given conditions. Some producers are defining l_n as the distance to the 0.2 m/s velocity contour in a height of 0.1 m above the floor. This will give an l_n of 0.35 m in the situation on the figure.

The velocity distribution downstream from three different wall-mounted diffusers is shown on figure 24. The maximum velocity u_x close to the floor (1 - 4 cm above the floor) is given as a function of the distance x from the diffuser.

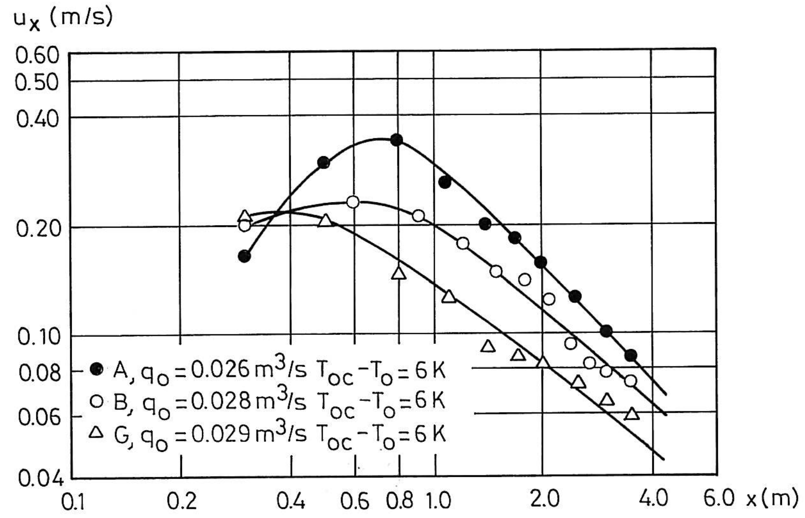

Figure 24. Maximum velocity close to the floor versus distance x. Reference /20/.

The cold air from supply opening A has a high initial acceleration due to buoyancy effect and a velocity of 0.34 m/s is obtained in a distance of 0.8 m from the diffuser. Type B has a large diffusion of the supply flow and the gravity will only increase the velocity to 0.23 m/s. The diffuser type G shows an even smaller velocity level although the flow to the room is almost the same in all three situtations but this diffuser has a higher velocity level outside the centre line.

Figure 24 indicates that the maximum velocity in the symmetry plane is proportional to $(1/x)^\alpha$ where the exponent α is close to 1.0 as pointed out by Nielsen et al. /16/.

It is obvious from figure 24 that different diffuser designs generate a different velocity level at the same flow rate and heat load.

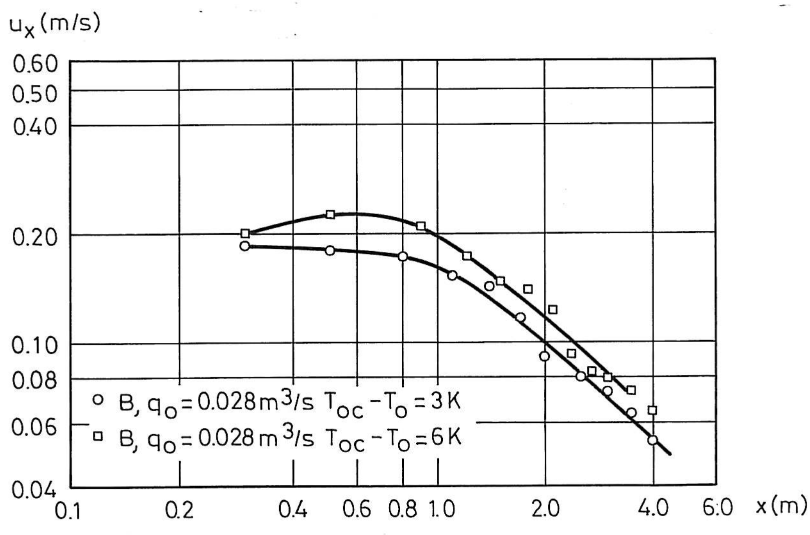

Figure 25. Velocity decay along the floor at different Archimedes' numbers. Reference /20/.

The velocity at the floor is not only influenced by the flow rate to the room and the type of diffuser. Figure 25 shows that the Archimedes number is an important parameter. A 3 °C increase in temperature difference will for example increase the velocity from 0.10 m/s to 0.12 m/s in a distance of 2 m from the diffuser. The figure shows that it is the gravity which accelerates the flow close to the diffuser resulting in a higher initial velocity level at higher Archimedes' numbers. This effect is very important for the flow in rooms with displacement ventilation and the outcome can be surprising. The velocity level in a room may for example be uninfluenced although the flow rate is reduced because the heat load in the room requires a reduction of the supply temperature and consequently an increase of the relative velocity level.

Profile measurements show that the flow in the vicinity of the floor can be characterized by a normalized velocity profile identical to the profile used for the description of wall jet flow, see reference /2/. The length scale δ in this profile is defined as the distance from the floor to the height where the velocity has a level which is half of the maximum velocity close to the floor, $u_x/2$.

Figure 26 shows the development in δ for three different Archimedes' numbers. (The characteristic temperature difference in the Archimedes number is in this chapter $T_{oc} - T_o$ where T_{oc} is the temperature in the height 1.1 m and the characteristic length is the height of the diffuser h). It can be seen that the height of the flow region is much smaller that the height of the diffuser, even at a distance of 0.5 m from the diffuser. The cold air from the diffuser accelerates towards the floor due to gravity and it behaves like a stratified flow in its further progress along the floor. δ is rather constant while it is proportional to x in a wall jet as indicated by the dotted line in figure 26. The length scale or thickness δ is slightly decreased at increasing Archimedes' number.

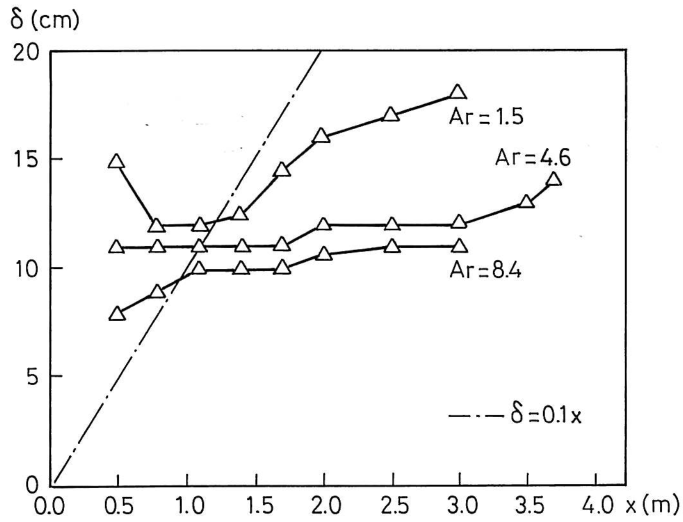

Figure 26. Length scale δ in the flow versus distance from the diffuser. Diffuser type G. Reference /20/.

The entrainment of air into the flow, or the turbulent mixing process, is diminishing when a vertical temperature gradient is present because the gravity will work against upward movement of heavy fluid and downward movement of light fluid. This is shown in hydraulics by for example Turner /21/ and it is shown for displacement ventilation by Jacobsen and Nielsen /2/.

The maximum velocity u_x in different distances from the opening x for a wall-mounted diffuser can be given by the following equation as shown by Nielsen /20/. The equation is based on stratified flow theory and it is confirmed by a number of measurements.

$$\frac{u_x}{u_f} = K_{dr} \frac{h}{x} \qquad (25)$$

h is the height of the diffuser and u_f is the face velocity defined as flow rate q_o divided by the face area a_f of the diffuser. K_{dr} is a function of the Archimedes number as well as an individual function for different types of air terminal devices. Both x and u_x are measured in the centre plane of the flow. The development of equation (25) assumes a high Archimedes number but the structure is also valid for cases where the Archimedes number is very small. In this case the flow will be a part of a potential core or a part of a radial wall jet. The velocity will in both cases be proportional to $1/x$ and equation (25) will therefore be able to predict the velocity u_x when the K_{dr}-value is adjusted to the situation.

The variables in equation (25) are easy to measure for a given diffuser and the equation is therefore simple to use in a practical design procedure.

Equation (25) can only be used at some distance from the diffuser as it appears from the figures 24 and 25. This distance is 1.0 m to 1.5 m for most of the diffusers. The equation will in any case give a velocity equal to or higher than the actual velocity and therefore a value which is suitable for a design procedure.

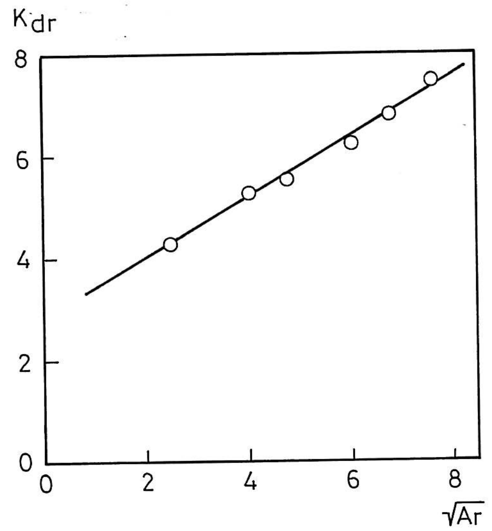

Figure 27. K_{dr} measured in the middle plane versus Archimedes' number for a wall-mounted air terminal device.

It is known from stratified flow in hydraulics that obstacles located downstream may influence the length scale δ of the flow, see reference /21/. Most of the measurements are made in test rooms of equal size so it is difficult to determine the influence from the end wall and the sidewalls, but practical experience indicates that room dimensions are of minor importance.

Figure 27 shows that K_{dr} increases with increasing Archimedes' number for the given diffuser. This is due to the fact that gravity will accelerate the vertical flow close to the opening and generate at stratified air movement in a relatively thin layer along the floor. Increased Archimedes' number will decrease δ and increase the maximum velocity in the layer. This effect is also shown on figure 25.

Mathisen /22/ has shown that the maximum velocity in the flow from a wall-mounted diffuser can be described as a linear function of $\sqrt{Ar}$. Figure 27 does confirm this assumption for large Archimedes' numbers, but deviations take place at smaller Archimedes' numbers for some products, reference /20/.

The flow from a single wall-mounted diffuser will be radial in the floor plane. The K_{dr}-values for different products will have different variations in directions outside the middle plane. It is typical that a new product has a small K_{dr} in the middle plane giving a small l_n, while older products have the largest K_{dr} in the middle plane, see /2/.

The flow from a number of diffusers placed close to each other on the wall will merge to a two-dimensional stratified flow. The velocity distribution can in certain areas be described by the equation

$$\frac{u_x}{u_f} = K_{dp} \tag{26}$$

if the Archimedes number is sufficiently large. It says that the velocity is independent of the distance but it is a function of flow rate, temperature difference and geometry around the diffusers. The same type of two-dimensional or plane flow will also take place in a narrow and a deep room with the diffuser located at the small end wall.

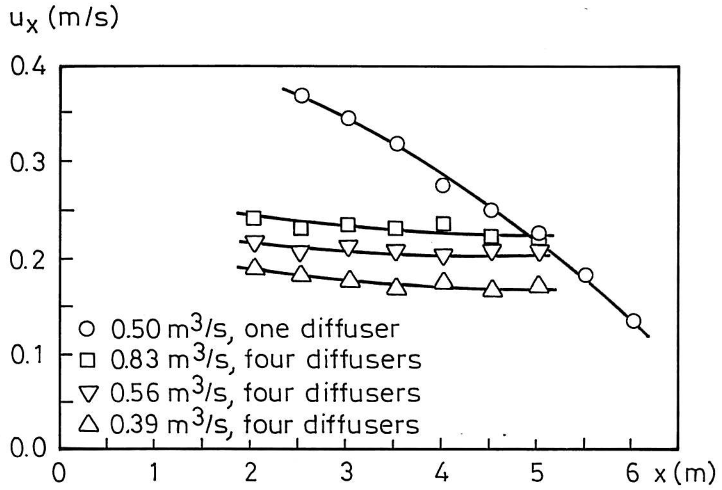

Figure 28. Velocity decay in radial flow from a single diffuser and velocity level in plane flow from a number of diffusers.

Nickel /23/ has measured the velocity distribution in the radial flow from a single diffuser. The diffuser is replaced by four diffusers located along one wall in a room with the width of 8 m. It is shown that two-dimensional flow gives the possibility to supply a high flow rate at a velocity level which is low compared to the velocity in the radial flow close to a single diffuser.

Velocity distribution in rooms with wall-mounted low velocity diffusers is also addressed by Mathisen /22/ and by Sandberg and Mattsson /24/. Fitzner /25/ shows full-scale measurements of stratified flow from sidewall-mounted diffusers and the interaction with a strong and a weak heat source in a typical office room.

5.2 Velocity Distribution from a Floor-Mounted Diffuser and from Diffusers Integrated into Furniture

Floor-mounted diffusers are often used in rooms with a high thermal load. The supply area is small and it is necessary with a supply velocity which is sufficient for a momentum driven flow in vertical direction (2 ~ 4 m/s).

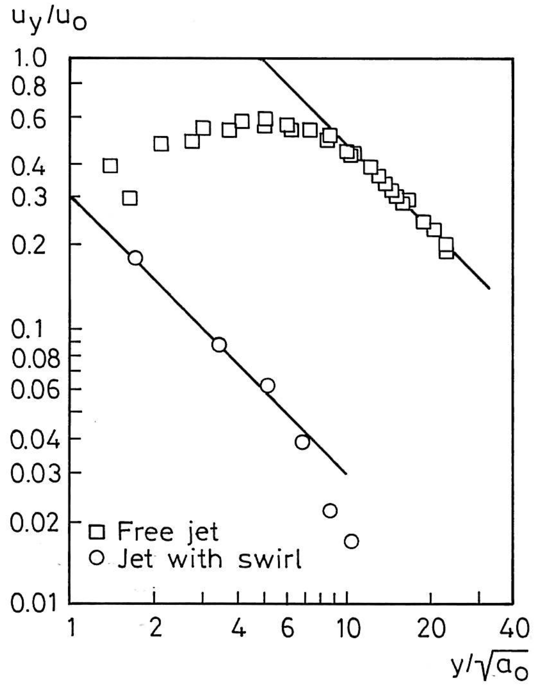

Figure 29. Velocity decay u_y/u_o in a free jet and in a jet with swirl versus the height y above the floor.

Figure 29 shows the velocity decay in free circular turbulent jet measured by Dittes and Mangelsdorf /26/. The openings can be located rather close in the floor dependent on the thermal load of the room. Entrainment into the jets will generate a recirculating flow above the opening and up to a height where the flow and temperature difference will dissolve the jets.

The velocity decay is given by the following equation for a free circular jet

$$\frac{u_y}{u_o} = \frac{K_a}{\sqrt{2}} \frac{\sqrt{a_o}}{y} \tag{27}$$

where u_y is the maximum velocity in distance y above the floor and a_o is the supply area of the diffuser. The K_a-value for the given diffuser is 6.8. A conventional air terminal device for the location in upper wall and ceiling in the case of mixing ventilation may have a K_a-value of the same level.

Figure 29 shows also measurements on a circular jet with swirl, reference /16/. It is convenient to compare the velocity decay in this jet with the velocity decay for a conventional free jet given by equation (27), and the velocity decay corresponds to a K_a-value of 0.42 which is about ten times smaller than the same value for a free jet. This is a surprising result and it shows that the swirl will generate a high entrainment and a very fast velocity decay. The diffuser if often used in a group of four within an area of 0.6 m × 0.6 m. The velocity level will in this case be higher than the velocity level from a single diffuser but both arrangements will have the same velocity level at a height of 0.8 m.

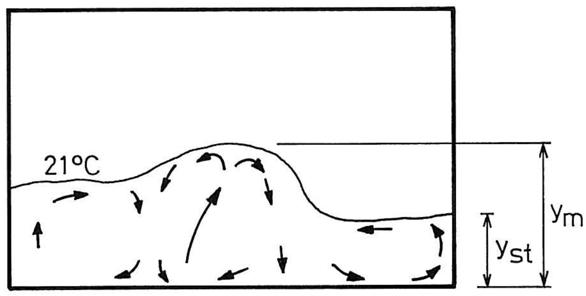

Figure 30. Flow in a room with a supply opening in the floor. Fitzner /10/.

The 21 °C isotherm in figure 30 indicates the boundary between lower and upper zones in a room with a floor-mounted supply opening. The recirculation around the cold jet generates a secondary flow in the lower part of the room with a movement towards the jet in a layer corresponding to the stratification height. The stratification height y_{st} is calculated from the entrainment into plumes in the room as discussed in chapter 3 and will not be influenced by the jet from the floor-mounted diffuser. The penetration height of the jet y_m is given by the following equation

$$\frac{y_m}{\sqrt{a_o}} = 3.33 \left(\frac{K_a u_o^2}{(T_{oc} - T_o)\sqrt{a_o}} \right)^{0.5} \tag{28}$$

when the supply temperature T_o is lower than the temperature in the occupied zone T_{oc}, reference /27/. y_m is the maximum penetration of the jet and this height is equivalent to the maximum penetration of a plume given by equation (8).

An integration of the supply opening in the furniture makes it possible to distribute the air efficiently in the occupied zone, especially in case of large rooms such as auditoria and lecture theatres. Figure 31 shows an air distribution system which is built into the seating structure. The conditioned primary air is fed from a pressure chamber accommodated in the chair mounting supports. Indoor air mixes with the primary air and the supply air emerges at the top of the back-rest, Rowlinson and Croome /28/.

The solution makes it possible to apply displacement ventilation in a large lecture theatre without having a draft risk. The recirculating room air flow that develops in downward systems is also avoided.

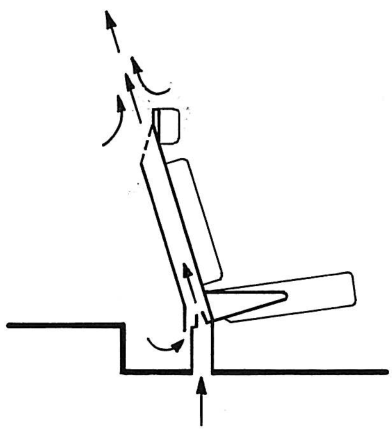

Figure 31. Seat with build-in supply system.

Typical system characteristics are a flow rate of 8 - 10 l/s per diffuser, a minimum supply temperature of 18 ^{0}C and a velocity of 1.5 - 2.5 m/s. The high location of the supply opening ensures a small temperature gradient in the occupied zone compared to the situation with supply openings at foot levels and the fresh air is supplied extremely close to the breathing zone of the people.

Supply openings can also be integrated into desks in an office. The supply system is connected via a double floor in the room. This system has the great advantage that it can be controlled partly by the individual users, reference /19/.

5.3 Flow between Obstacles in the Occupied Zone and Flow from Cold Downdraft

The flow in the vicinity of the floor may be influenced by furniture and by other obstacles in the occupied zone. The maximum velocity in the flow is located rather close to the floor (between 1 to 4 cm above the floor), and a great deal of the air movement will therefore take place in this region. Conventional furniture will only have a small influence on the air movement while obstacles placed direct on the floor will block the flow. An opening between this type of obstacles will work as new supply opening because the flow in the room is stratified. Figure 32 shows the flow between two obstacles where the cold air is supplied in the left side of the room and the heat sources are located in the right side of the room.

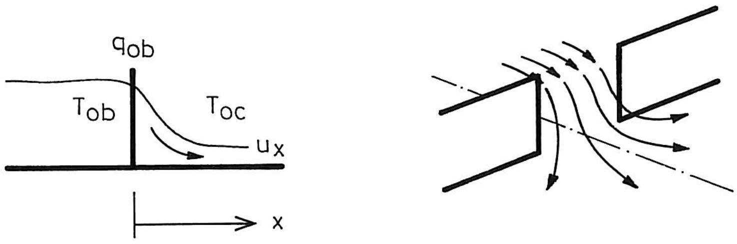

Figure 32. Radial stratified flow between obstacles.

Nielsen /20/ has shown that the flow from an opening between obstacles can be described as a semi-radial flow like the air movement from a wall-mounted supply opening. The velocity decay can be described by the equation

$$\frac{u_x}{q_{ob}} = K_{ob} \frac{1}{x} \tag{29}$$

where u_x is maximum velocity in distance x from the opening and q_{ob} is the excess air supplied on the upstream side. u_x is measured in the symmetry plane.

Figure 33 shows the measurements of K_{ob} in equation (29). The structure of equation (29) and the distribution of K_{ob}-values are equivalent to the structure of equation (25) and the distribution of K_{dr}-values in figure 27.

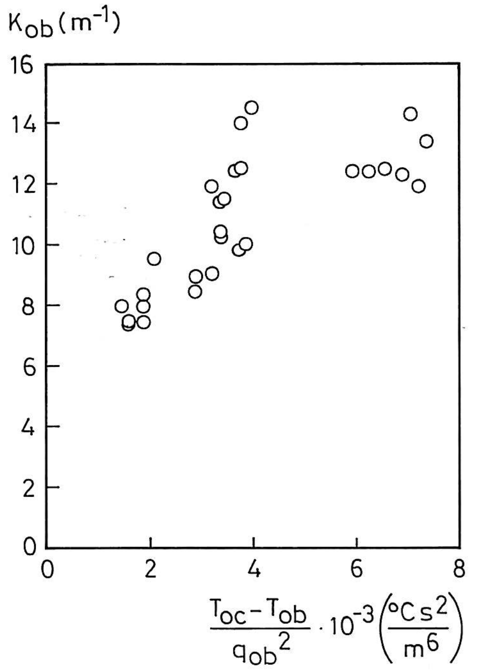

Figure 33. K_{ob} versus flow rate and temperature difference. Reference /20/.

The temperature difference $T_{oc} - T_{ob}$ is the difference between the temperature in the height 1.1 m in front of the opening and the lowest temperature in the opening between the obstacles. q_{ob} is the flow rate between the obstacles. The width of the opening is varying from 0.1 m to 1.5 m in the experiment on figure 33. The measurements show that the importance of the width is less obvious and results for different widths are given on the figure.

Cold downdraft from surfaces can be another reason for a velocity level in the occupied zone. The maximum velocity in a cold downdraft is given by /29/.

$$u_y = 0.055 (y \Delta T)^{0.5} \tag{30}$$

where y is height of the cold surface and ΔT is the temperature difference between room temperature and surface temperature.

The cold downdraft will be deflected into the occupied zone at the floor and it will move as a stratified flow with a typical wall jet profile /29/. Two flow domains can be identified. One domain with growth of thickness (length-scale) which indicates entrainment of room air into the cold airflow and one with a constant thickness and a constant velocity which is typical of a plane stratified flow similar to the flow given by equation (26).

The maximum velocity in plane flow along the floor can be given by the folowing equation /29/.

$$u_x = 0.094 \frac{\sqrt{y \Delta T}}{x + 1.31} \qquad 0.4 \le x \le 2.0 \tag{31}$$

$$u_x = 0.028 \sqrt{y \Delta T} \qquad x > 2.0 \tag{32}$$

u_x is the maximum velocity close to the floor at distance x and y is the height of the cold surface. ΔT is temperature difference between room temperature and surface temperature. The measurements are made in rooms with a length of 3 to 7 m.

The flow along the floor will be radial if the cold surface is narrow compared to the width of the wall. The velocity will decrease rapidly from the level given by (30) in a velocity decay similar to the decay given in equation (25) and (29).

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LISTS OF SYMBOLS

Ar	Archimedes' number	
a_f	Face area	m^2
a_o	Diffuser supply area	m^2
c	Concentration	mg/m^3 or cm^3/m^3
c_{oc}	Mean concentration in occupied zone	mg/m^3 or cm^3/m^3
c_P	Concentration in point P	mg/m^3 or cm^3/m^3
c_R	Concentration in return opening	mg/m^3 or cm^3/m^3
$\bar{c}$	Mean concentration in the room	mg/m^3 or cm^3/m^3
c_1	Concentration in the lower zone	mg/m^3 or cm^3/m^3
d	Hydraulic diameter of heat source	m
g	Gravitational acceleration	m/s^2
H	Height of room	m
h	Height of wall-mounted diffuser	m
K_a	Constant for circular free jet	
K_{dp}	Coefficient for plane flow	
K_{dr}	Coefficient for radial flow	
K_{ob}	Coefficient for flow between obstacles	m^{-1}
k	Coefficient for convective heat emission	
l	Length of heat source. Width of surface	m
l_n	Length from diffuser to 0.2 m/s velocity contour	m
m	Dimensionless parameter	
N	Number of heat sources	
q_y	Volume flow in thermal plume or cold downdraft	m^3/s
q_{ob}	Volume flow between obstacles	m^3/s
q_{yN}	Volume flow from N heat sources	m^3/s
q_{yc}	Volume flow from a heat source located in a corner	m^3/s
q_{yw}	Volume flow from a heat source close to a wall	m^3/s
q_o	Volume flow supplied to the room or flow rate from a diffuser	m^3/s
q_1, q_2, q_3	Volume flows in plumes and downdraft	m^3/s
T	Temperature	°C
T_f	Minimum temperature close to floor	°C
T_{ob}	Minimum temperature in flow between obstacles	°C
T_{oc}	Mean temperature in occupied zone or temperature in the height of 1.1 m	°C
T_R	Return temperature	°C
T_o	Supply temperature	°C
u_f	Face velocity. Flow rate divided by face area	m/s
u_x	Maximum velocity at distance x	m/s

u_y	Maximum velocity at height y	m/s
u_o	Supply velocity	m/s
x	Coordinate and distance	m
y	Coordinate, distance and length of surface	m
y_m	Maximum height of a plume and a cold jet	m
y_t	Height to buoyancy neutral flow	m
y_{oc}	Height of the occupied zone	m
y_{st}	Stratification height	m
y_o	Height from virtual origin to heat source reference level	m
y^*	Dimensionless height	
α	Exponent	
β	Volume expansion coefficient	K^{-1}
ΔT	Temperature difference between room temperature and surface temperature	K
ΔT_o	Temperature difference $T_R - T_o$	K
δ	Thickness or length scale of flow	m
ε_P	Ventilation index	
ε_T	Temperature effectiveness	
ε_{oc}	Ventilation effectiveness of the occupied zone	
$\overline{\varepsilon}$	Mean ventilation effectiveness of the room	
μ_T	Load factor	
ϕ	Heat emission	W
ϕ_K	Convective heat emission	W